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ABSTRACT

Demands for higher energy efficiencies in both residential and commercial refrigeration and air conditioning systems have resulted in a trend toward heat exchanger designs that are more compact with higher capacities for heat transfer. Traditional copper tube/aluminum fin coil manufacturing technology remains prevalent throughout the industry and, when modified for smaller diameter copper tubes of 5mm or less, significant improvements in heat transfer can be achieved. When coupled with internal enhancements to the copper tubes such as microgrooves, coil designs can be smaller, more efficient and less costly.

Using a commercially available heat exchanger design and simulation software and CFD modeling, this paper compares optimized 3-ton air conditioning condenser coils manufactured with small-diameter internally enhanced copper tubes against condensers with aluminum microchannel tubes. Simulated operating conditions are held constant, including refrigerant inlet pressure and temperature, as well as air flow rate and inlet temperature. Comparisons of material consumption, refrigerant charge, volume, and heat transfer performance are demonstrated. It was found that using internally enhanced copper tubes with a diameter of 5mm, condenser coils can be designed to operate with less refrigerant charge and have the potential to be lighter and more compact than commercially available, optimized aluminum coil designs with microchannel tubes.

1. INTRODUCTION

Regulations related to energy efficiency and refrigerants that are less damaging to our atmosphere have resulted in demands for more efficient, more compact heat exchangers that can operate at higher refrigerant pressures. Newer, optimized coil designs have been meeting these demands by using either smaller diameter round copper tubes or aluminum microchannel tubes. Dramatically different coil manufacturing processes are required for aluminum microchannel tubes compared to round copper tubes, and discussions among air conditioning and refrigeration professionals continue to debate the benefits of one technology over the other.

Traditional copper tube/aluminum fin coil manufacturing technology remains prevalent in the industry and in recent years, a trend has emerged toward smaller diameter tubes. Published studies and documented trials (You, 2011) have demonstrated benefits from smaller diameter tubes including: increased performance, higher system efficiencies, lower refrigerant charge and material savings. A considerable amount of work related to 5mm copper tubes in window and split-type air conditioning units has been performed in China at the Institute of Refrigeration and Cryogenics, Shanghai Jiao Tong University. Ding et al (2010) reported refrigerant reductions of 20% to 30% when reducing condenser tube diameters from 7mm to 5mm, but noted higher refrigerant-side pressure drops with the 5mm tubes as a potential issue that can be addressed with increased tube circuits.

Filippini et al (2010) reported optimized heat exchanger designs with 5mm copper tubes and also with 30mm aluminum microchannel tubes, both optimized designs intended as a replacement for a baseline condenser coil made with traditional 9.52mm copper tubes. In the Filippini study (2010), refrigerant charge in the microchannel heat exchanger was reported at 50.9% of the baseline 9.52mm coil and the refrigerant charge in the design with 5mm copper tubes was 43.6% of the baseline. Although all the heat exchangers in the Filippini study (2010) had similar cooling capacity, frontal areas and fans, the microchannel tubes were 30mm x 2mm and it could be argued that these microchannel tubes did not represent current state-of-the-art in aluminum heat exchanger technology. The study reported in this paper attempts to maintain similar operating conditions and also compare state-of-the-art technology between heat exchangers made with microchannel aluminum tubes and 5mm copper tubes.

The overall goal of this paper is to provide a meaningful comparison of coils with aluminum microchannel tubes and 5mm enhanced copper tubes. This study was designed to meet the following objectives:

- Design and optimize an air conditioning coil using 5mm enhanced copper tubes based on true and consistent operating conditions such as saturation pressure, subcooling, air flow rate and inlet temperature, and fin geometry
- Compare the 5mm round copper tube and multiport extruded aluminum tube heat exchangers for their material consumption and refrigerant charge maintaining the same heat transfer performance and comparable system energy efficiency.

It is known that with round tubes, a wide variety of circuitry options are available, such as counter flow configurations, optimization of mass flux along refrigerant flow direction through tube merging or splitting, and elimination of detrimental tube or fin heat conductions. Circuitry options for multichannel aluminum tubes are considerably limited in comparison to round tubes. In this study, multiple refrigerant circuits were considered in the 5mm optimized designs, resulting in improved efficiency and refrigerant pressure drops equal to or less than the baseline aluminum tube heat exchanger.

2. Design and Optimization of 5mm Tube Condenser Coils

Air conditioning is a major application of refrigerant to air heat exchangers, so it was determined that an analysis of 3-ton AC condenser coils could provide objective and meaningful comparison between microchannel heat exchangers and 5mm copper tube and aluminum fin heat exchangers. A representative baseline microchannel coil was selected and modeled and then an exhaustive search optimization was carried out to analyze the performance of 3-ton AC condenser coils throughout a large design space.

2.1 Baseline Microchannel Coil

The baseline aluminum microchannel condenser coil was purchased from a local HVAC contractor supply house. This coil is a specified OEM replacement for a commercially available, residential 3-ton, 13 SEER central AC outdoor condenser unit. The microchannel tubes in this coil are 18mm wide x 1.3mm high with 23 channels in each tube. Channel dimensions are 0.53mm x 0.82mm. It was concluded that this coil represents the current state-of-the-art in aluminum microchannel condenser coils for 3-ton central AC units. The manufacturer-rated heat rejection for this coil is 13,400 watts with R410A refrigerant at 2.78 MPa inlet and 5.9K subcooling.

2.2 Copper Tube and Aluminum Fin Selection and Analysis

A commercially available copper tube with internal microgrooves and a 5mm outer diameter was selected for modeling the proposed heat exchanger designs. The tube manufacturer provided test data for heat transfer coefficient and refrigerant pressure drop. An analysis determined that the existing correlations, including the Shah correlation (Shah, 1979) for condensation heat transfer, Gnielinski correlation (Gnielinski, 1976) for single-phase heat transfer, Friedel correlation (Friedel, 1979) for two-phase pressure drop and Blasius equation (DeWitt, 1996) for single-phase pressure drop, without appropriate correction factors, can predict the manufacturer data accurately.

A slit fin design with 1.0mm slit width and 0.9mm slit height was selected for the heat exchangers to be evaluated. These enhanced fins offer better heat transfer performance than flat plate fins and are employed in similar heat exchangers. A detailed CFD analysis was performed to understand the heat transfer and pressure drop performance of these fins. The CFD work is described in section 3.5 and was used to determine the appropriate heat transfer and pressure drop correlations and correction factors for the simulation of coils with these fins.

2.3 Modeling Approach

A commercially available heat exchanger design and simulation software package was used to model the performance of heat exchangers designed in this study. The software is capable of simulating tube fin heat exchangers as well as microchannel heat exchangers. The software includes the aforementioned Shah, Gnielinski, Friedel, and Blasius correlations as well as the Wang-Lee-Sheu (2001) slit-fin correlations. Coils designed in this software can essentially have any arrangement of tubes, fins, and refrigerant, so this allowed us to explore a very large design space by building and solving coils of many different configurations.

The software was first applied to model the baseline microchannel condenser, and the results matched the experimental data well.

2.4 Optimization Methodology

To find the best designs for a 3-ton condenser using 5mm copper tubes, an exhaustive search optimization was performed to find the optimum designs within a finite set of parameters. To perform the optimization, a custom application was written in the C# programming language to "build" and calculate the performance of these coils with the heat exchanger design software previously mentioned. The code generates tens of thousands of heat exchanger designs by varying all combinations of design parameters specified. Each of these designs is simulated in the software to determine the performance of the heat exchanger based on specific inlet conditions.

In order to develop a comprehensive understanding of the design space for 5mm copper tube and aluminum fin coils, an exhaustive search was performed by varying the values of key design parameters. All combinations of the following parameters were designed, solved, and analyzed:

- Fins per Meter (or Fins per Inch): 630 (16), 709 (18), 789 (20), 866 (22), 945 (24) and 1,024 (26)
- Horizontal Tube Spacing (x tube OD): 1.5, 1.75, 2, 2.25, 2.5, 2.75 and 3
- Vertical Spacing (x horizontal spacing): 1, 1.25, 1.5, 1.75, 2 and 2.25
- Tubes per Row: 16, 20, 24, 28, 32, 36, 40, 44 and 48
- Tube Rows: 1, 2
- Tube Lengths (m): 1.25, 1.5, 1.75, 1.943, 2 and 2.25
- Number of Refrigerant Circuits: (tubes per row)/2, (tubes per row)/4; (tubes per row)/6, and (tubes per row)/8 for designs which are evenly divisible only

All coils were simulated under identical conditions as follows; they are based on realistic condenser inlet conditions from a representative vapor compression cycle.

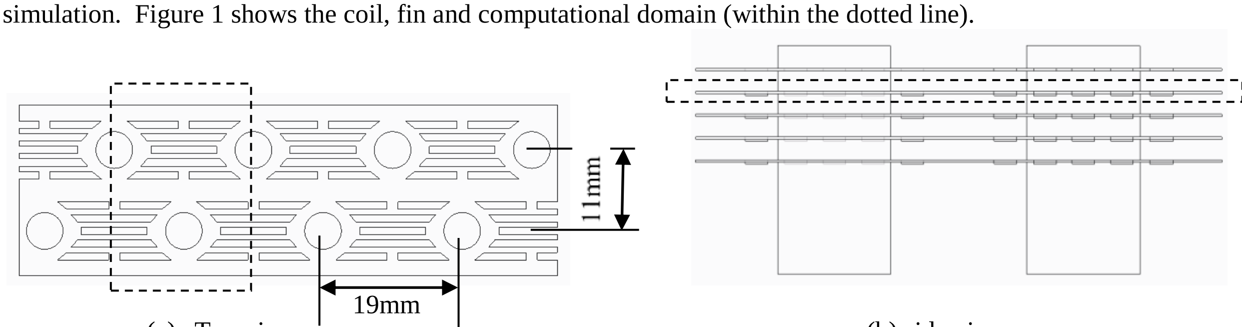
- Air entering temperature: 35°C
- Air flow rate: 1.274 m^3/s
- Refrigerant Inlet Pressure: 278 M-Pa
- Refrigerant Inlet Temperature: 72.6°C
- Refrigerant Mass Flow Rate: 0.06615 kg/s

To ensure the same performance to the baseline microchannel coil, the raw data from the exhaustive search was filtered to allow an airside pressure drop no greater than the 115% of the baseline pressure drop (Max airside pressure drop = 18.45 Pa). Additionally the refrigerant outlet temperature was limited to ±0.15°C of the baseline outlet temperature of 38.16°C. Finally, results with low performance (Capacity <13,000W) were removed from the results. Accepting only results with nearly identical outlet conditions and performance ensures that the comparison is fair and the performance of the rest of the vapor compression cycle is not affected by the new heat exchanger.

2.5 CFD Simulation for 5mm Tube Coil

The correlation developed by Wang, Lee and Sheu (Wang, 2001) has been commonly used in the industry to predict heat transfer performance for slit fin coils, but the database for developing the correlation only includes 7.52 – 16.4mm tube outside diameter. To verify that the Wang-Lee-Sheu correlation is applicable to the 5 mm tube and slit fin coils in our investigation, CFD simulation has been conducted for a two-row 5 mm tube coil to obtain a numerical prediction of heat transfer coefficient and pressure drop and compare to the Wang-Lee-Sheu correlation.

2.5.1 Numerical Method: Due to periodic boundary conditions, only one piece of fin was needed for the CFD simulation. Figure 1 shows the coil, fin and computational domain (within the dotted line).



(a) Top view (b) side view

Figure 1: Coil sketch and computational domain

The coil has a 19 x 11 mm tube pattern. The slit fin, or more specifically offset-strip fin in this case, has a 5-slit set with 1.0 mm slit breadth and 0.9 mm slit height. The domain was discretized with hexahedral meshes (totally 1.1 million grids). The Navier-Stokes equations and energy equation were solved using a commercially available CFD code. Due to low velocity and small fin pitch, the flow was assumed to be laminar and steady. Pressure-velocity coupling was performed with SIMPLE scheme, and second order upwind discretization was implemented with the momentum and energy equation. The data reduction method is as follows:

$$h_o = \frac{Q}{\eta A_o \Delta T} \tag{1}$$

$$\Delta T = \frac{(T_{in}-T_t)-(T_{out}-T_t)}{\ln\frac{T_{in}-T_t}{T_{out}-T_t}} \tag{2}$$

Where h_o is air side heat transfer coefficient, η is surface efficiency, Q is heat transfer rate, A_o is total heat transfer area, T_{in} and T_{out} are air inlet and out temperature (mass averaged), respectively, and T_t is the tube surface temperature.

The dimensionless parameters, Colburn j-factor, Fanning fraction factor, Reynolds number, are defined as follows:

$$j = \left(\frac{h_o}{\rho V_c c_p}\right) \cdot Pr^{2/3} \tag{3}$$

$$f = \frac{\Delta P}{\frac{A_o}{A_c}\frac{\rho V_c^2}{2}} \tag{4}$$

$$Re = \frac{\rho V_c D_o}{\mu} \tag{5}$$

Where V_c is the mean velocity at the minimum flow cross-section, ΔP is the pressure drop across the fin-core in the computational domain, A_c is the minimum flow cross-section area, D_o is the tube outside diameter, ρ is air density, μ is dynamic viscosity, c_p is specific heat, and Pr is the Prandtl number.

2.5.2 Numerical Results: In the CFD work, two different fin densities, 23 fins/in (fin pitch of 1.1 mm) and 15 fins/in (fin pitch of 1.693 mm), were simulated. Air inlet temperature and tube surface temperature were set at 35°C and 48.9°C, respectively, and various frontal air velocities between 0.75 m/s and 4.5 m/s were studied. Figure 2 shows airflow velocity vectors on one cross-section between fins at 1 m/s of frontal velocity and 1.1 mm of fin pitch, and Figure 3 shows the temperature distribution for the same location. The wake area behind the 5 mm tubes is obviously smaller than larger diameter tube coils studied in previous CFD work (Zhang et al., 2000). This can increase heat transfer and lower pressure drop.

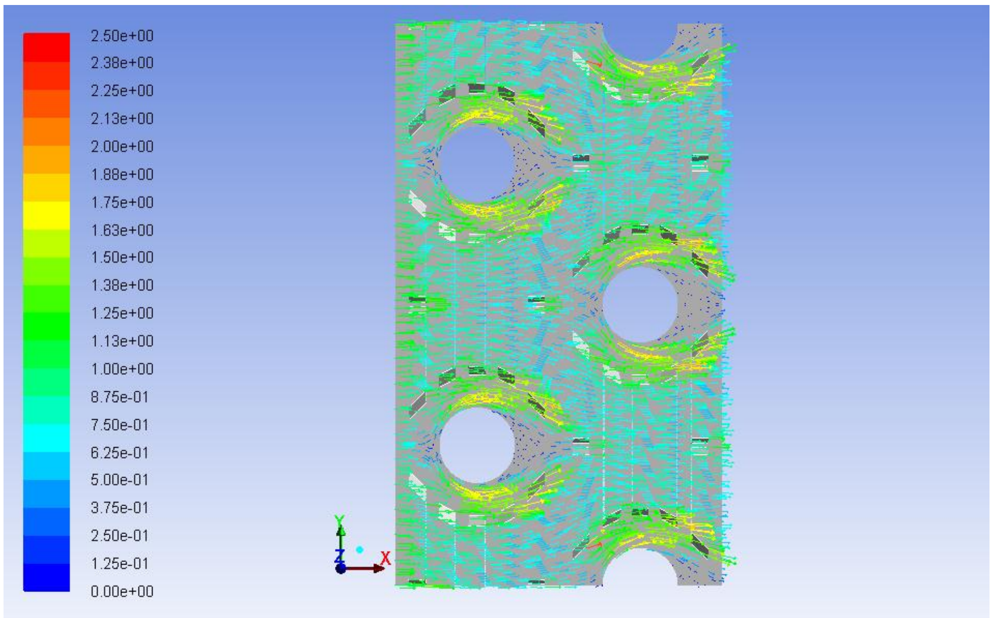


Figure 2: Local velocity distributions (velocity vector colored by velocity magnitude)

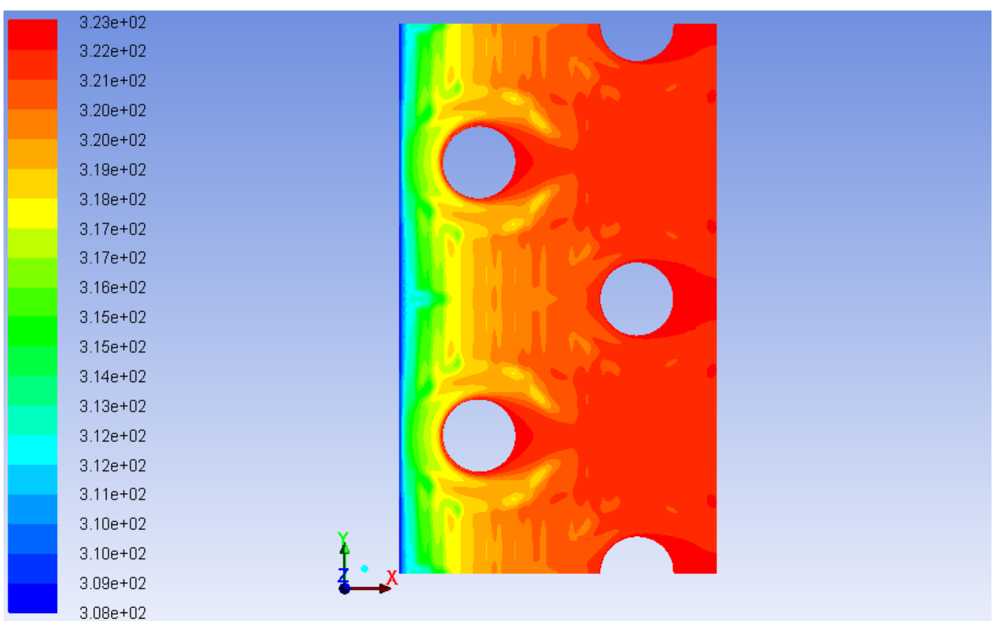


Figure 3: Air temperature distribution

Figure 4 presents heat transfer coefficient at various frontal velocities and Colburn-j factor vs. Reynolds number. Figure 5 presents pressure drop vs. frontal velocities and friction factor vs. Reynolds number. One can see for the two-row case the CFD results match the Wang-Lee-Sheu correlation well except for velocity lower than 1 m/s. However, the CFD-predicted pressure drop is consistently higher than that predicted by Wang-Lee-Sheu correlation. For 1.1 mm fin pitch, the difference in friction factors by the two methods is approximately 18% at 1 m/s and 32% at 4.5 m/s. Considering the frontal velocity was between 1 and 2 m/s for most calculations in the design and optimization of the 5 mm tube coils, we used a correction factor of 1.2 for pressure drop calculation using the Wang-Lee-Sheu correlation.

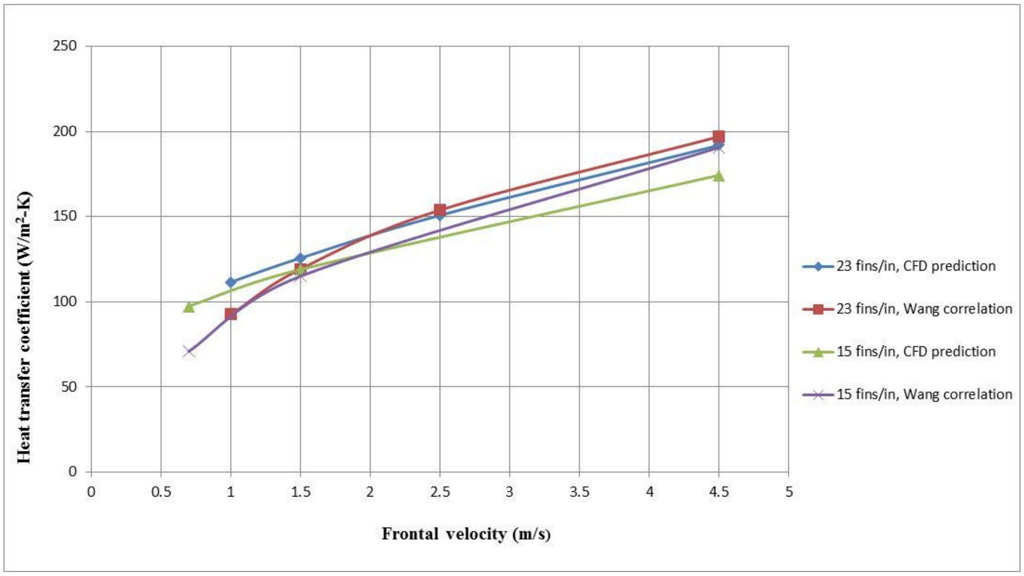


Figure 4(a): Heat Transfer Results for 1.1mm fin pitch – Heat transfer coefficient vs. frontal velocity

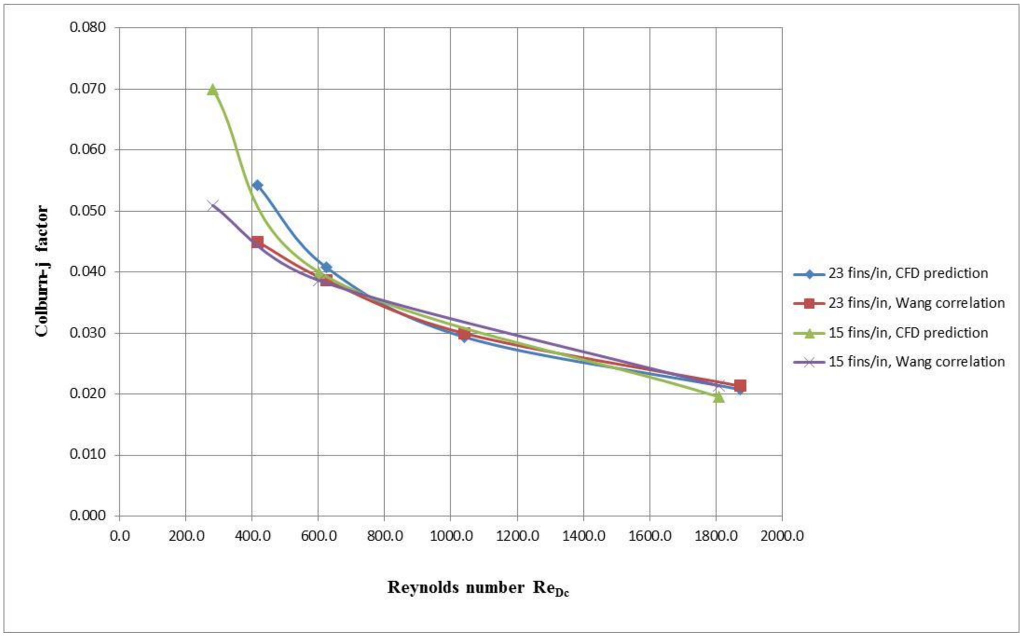


Figure 4(b): Heat transfer results for 1.1 mm fin pitch – Colburn j factor vs. Reynolds number

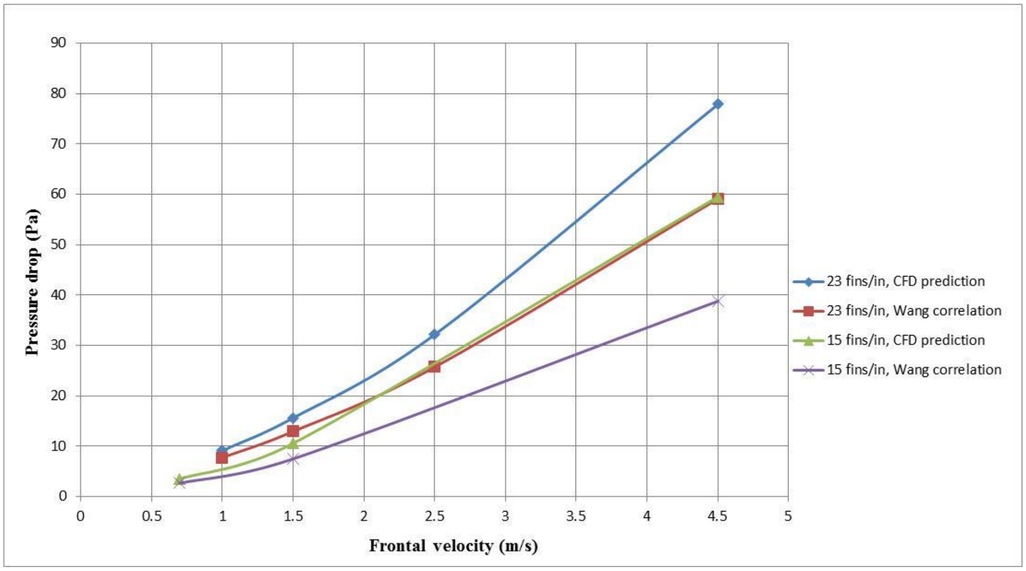


Figure 5(a): Pressure drop results for 1.1 mm fin pitch – Pressure drop vs. Frontal velocity

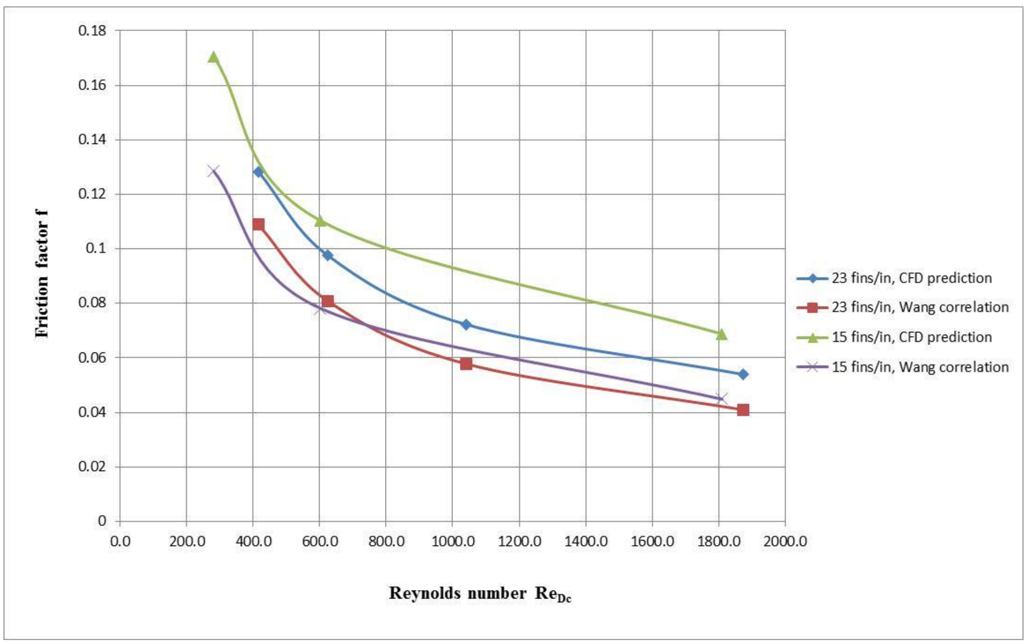


Figure 5 (b): Pressure drop results for 1.1 mm fin pitch – Friction factor vs. Reynolds number

3. RESULTS AND DISCUSSION

After the optimization program generated and simulated all of the coils within the design space, designs with poor performance were removed. The filtered data was entered into data visualization software to compare the designs based on several different parameters including coil volume, refrigerant charge, and material mass. The software is able to output the Pareto optimum points, which are the optimal combinations of the two objectives given. This section highlights some designs of interest for each of the objectives.

3.1 Minimizing Coil Volume

Figure 6 shows heat rejection rates versus coil volumes for all the design points meeting performance requirements. Each point in the figure represents one coil design. The Pareto optimum points (blue) were identified for the various coil volumes versus the heat rejection of the filtered data. The coil design with the minimum volume (run #20914) and its comparison to the baseline microchannel are given in Table 1. This coil has a volume of 0.0193 m^3, or 90.6% of the baseline.

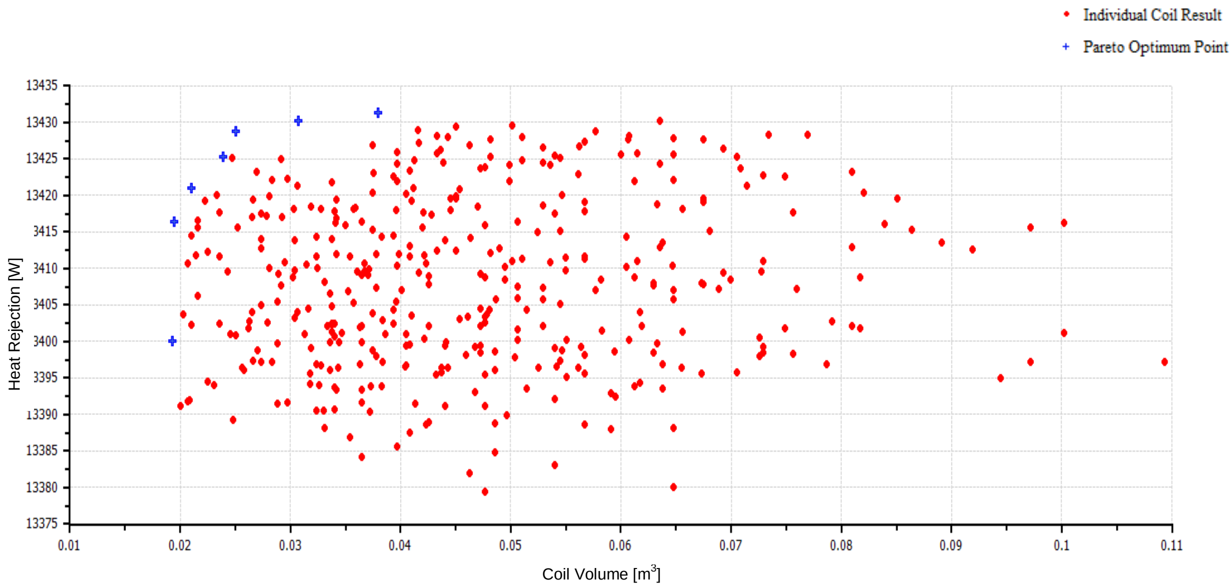


Figure 6: Heat Rejection Rates vs. Coil Volumes

3.2 Minimum Refrigerant Charge

Figures 7 shows heat rejection versus refrigerant charge with Pareto optimum points in blue. The design with the least charge (run #38196) has 0.46 kg of refrigerant, about 80% of the baseline (see Table 1).

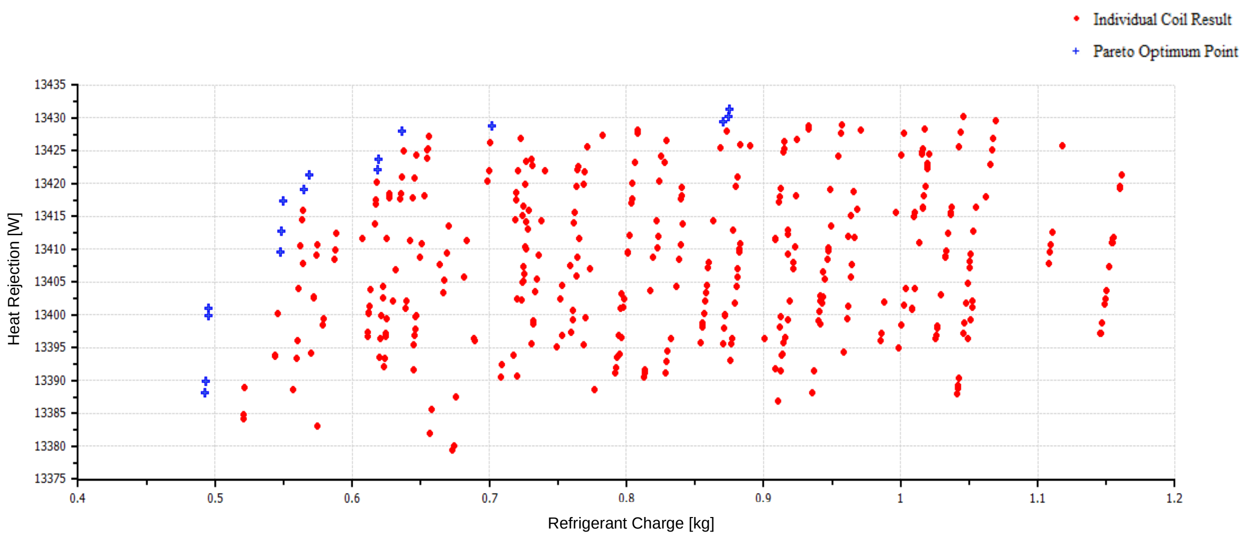


Figure 7: Heat Rejection vs. Refrigerant Charge

3.3 Minimizing Material Consumption

Figure 8 shows the mass of fin material versus the mass of tube material. The lightest design (run #56708) has a total mass of 10.56 kg; about 1.52 kg, or 16.8% more than the baseline aluminum coil (see Table 1). Note that the

5mm tubes in this study are commercially available and have a tube wall thickness of 0.21mm. These tubes have a burst pressure above 18,616 kPa (2,700 psi) and meet the Underwriter Labs (UL) burst pressure standard of 5 times the operating pressure. The aluminum tube burst pressure is above 13,790 kPa (2,000 psi). With newer refrigerants that operate at higher pressures, manufacturers are beginning to use the UL accepted alternate of 3 times the operating pressure plus a fatigue test. It is estimated that with a 5mm tube wall thickness of 0.17mm, the burst pressure specification of 3 times operating pressure could be met and the overall mass of the simulated coil reduced, resulting in total coil weights that are within 2% to 3% of each other.

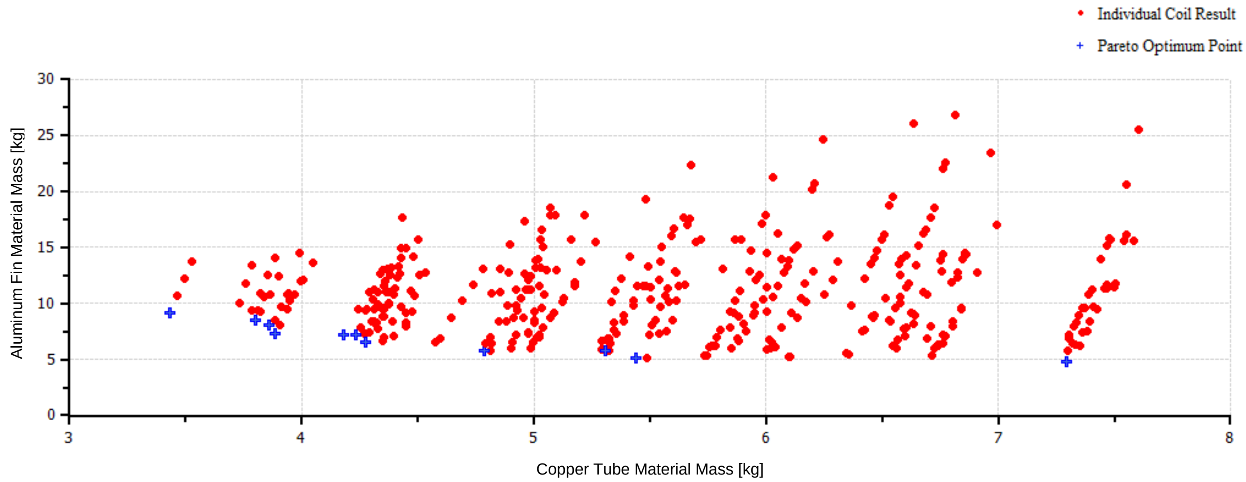


Figure 8: Fin Material Mass vs. Tube Material Mass

Table 1: Coil Design with Minimum Volume, Refrigerant Charge and Material Consumption and Their Comparison to the Baseline

	Baseline	5mm Run #29014 (Minimum Volume)	5mm Run #38196 (Minimum Refrigerant)	5mm Run #56708 (Minimum Coil Mass)
Tube Length (m)	1.943	1.750	1.750	2.000
Tube Spacing – Horizontal (m)		0.0088	0.015	0.01
Tube Spacing – Vertical (m)	0.0127	0.0131	0.0225	0.0125
Circuits		12	7	10
Tubes per Bank	47	48	28	40
Number of Tube Banks (Rows)	1	2	2	2
Fin Density: Fins per meter (Fins per Inch)	906 (23)	1024 (26)	1024 (26)	945 (24)
Air Pressure Drop (Pa)	16.05	13.23	16.24	16.8
Refrigerant Pressure Drop (Pa)	108,081.7	40,216.9	102,430.7	63,022.1
Coil Volume (m^3)	0.0213	0.0193	0.0331	0.0200
Face Area (m^2)	1.1819	1.1025	1.1025	1.0000
Heat Rejection (W)	13,406	13,400	13,388	13,391
Subcooling (K)	5.90	6.86	5.82	6.44
Refrigerant Exit Temperature (K)	311.3	311.4	311.4	311.4
Copper Mass (kg)		5.75	3.43	5.46
Aluminum Mass (kg)	9.04	5.33	9.14	5.10
Refrigerant Charge (kg)	0.57	0.77	0.46	0.72

4. CONCLUSIONS

Based on computer simulated results for a residential 3-ton AC condenser coil, and compared to a commercially available coil with 18mm x 1.3mm microchannel aluminum tubes, coils could be manufactured with 5mm internally enhanced copper tubes with the following results:

- Minimized Coil Volume – The 5mm coil design with the least volume is 90.7% of the volume of the baseline coil.
- Refrigerant Charge - The 5mm coil design with the least refrigerant charge is approximately 80% of the refrigerant charge required in the baseline coil.
- Material Consumption - The 5mm coil design with the least mass is approximately 16.8% higher than the baseline coil. However, thinner tube walls will meet UL standards and reduce the coil mass such that it is within 2 to 3% of the baseline aluminum microchannel coil. (You, 2011)

NOMENCLATURE

A	heat transfer surface area	(m^2)	j	Colburn j-factor
h	heat transfer coefficient	($W/(m^2K)$)	f	Fanning friction factor
P	pressure	(kPa)	Re	Reynolds number
Q	heat transfer	(W)	Pr	Prandtl number
D	diameter	(m)	**Subscripts**	
T	temperature	(K)	c	minimum flow cross section
V	velocity	(m/s)	in	tube inlet
c	specific heat capacity	(kJ/kg-K)	out	tube outlet
μ	dynamic viscosity	(kg/m-s)	o	overall/ outside
ρ	density	(kg/m^3)	t	tube surface
η	surface efficiency	-	p	constant pressure

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